

NASA-CR-191792

SBIR 08.12-3800
Release Date-4/05/94



(NASA-CR-191792) THE 4 K STIRLING
CRYOCOOLER DEMONSTRATION Final
Report No. 8, 1 May 1990 - 30 Apr.
1992 (Creare) 38 p

N94-32484

Unclas

G3/31 0010555



Engineering Research and Development
Computer Systems and Software

TECHNICAL MEMORANDUM

4 K STIRLING CRYOCOOLER
DEMONSTRATION

TM-1563

SEPTEMBER 1992

NA 57-1111

TECHNICAL MEMORANDUM

4 K STIRLING CRYOCOOLER DEMONSTRATION

W. Dodd Stacy
Creare Inc.
Hanover, NH

For

NASA-JET PROPULSION LABORATORY
Contract No. NAS7-1111
May 1, 1990 - April 30, 1992

SBIR Rights Notice (June 1987)

~~These SBIR data are furnished with SBIR rights under Contract No. NAS7-1111. For a period of 2 years after acceptance of all items to be delivered under this contract, the Government agrees to use these data for Government purposes only, and they shall not be disclosed outside the Government (including disclosure for procurement purposes) during such period without permission of the Contractor, except that, subject to the foregoing use and disclosure prohibitions, such data may be disclosed for use by support Contractors. After the aforesaid 2-year period the Government has a royalty-free license to use, and to authorize others to use on its behalf, these data for Government purposes, but is relieved of all disclosure prohibitions and assumes no liability for unauthorized use of these data by third parties. This Notice shall be affixed to any reproductions of these data, in whole or in part.~~

EXECUTIVE SUMMARY

This report briefly summarizes the results and conclusions from an SBIR program intended to demonstrate an innovative Stirling cycle cryocooler concept for efficiently lifting heat from 4 K. Refrigeration at 4 K, a temperature useful for superconductors and sensitive instruments, is beyond the reach of conventional regenerative thermodynamic cycles due to the rapid loss of regenerator matrix heat capacity at temperatures below about 20 K. To overcome this fundamental limit, the cryocooler developed under this program integrated three unique features: recuperative regeneration between the displacement gas flow streams of two independent Stirling cycles operating at 180° phase angle, tailored distortion of the two expander volume waveforms from sinusoidal to perfectly match the instantaneous regenerator heat flux from the two cycles and thereby unload the regenerator, and metal diaphragm working volumes to promote near isothermal expansion and compression processes. Use of diaphragms also provides unlimited operating life potential and eliminates bearings and high precision running seals. A Phase I proof-of-principle experiment demonstrated that counterflow regenerator operation between 77 K and 4 K increases regenerator effectiveness by minimizing metal temperature transient cycling. In Phase II, a detailed design package for a breadboard cryocooler was completed. Fabrication techniques were successfully developed for manufacturing high precision miniature parallel plate recuperators, and samples were produced and inspected. Process development for fabricating suitably flat diaphragms proved more difficult and expensive than anticipated, and construction of the cryocooler was suspended at a completion level of approximately 75%. Subsequent development efforts on other projects have successfully overcome diaphragm fabrication difficulties, and alternate funding is currently being sought for completion and demonstration testing of the 4 K Stirling cryocooler.

TABLE OF CONTENTS

EXECUTIVE SUMMARY.....	i
TABLE OF CONTENTS.....	ii
LIST OF FIGURES.....	iii
I. INTRODUCTION.....	1
I.1 4 K Cooler Concept.....	1
I.2 Phase I Objective.....	2
I.3 Phase II Objective.....	2
II. SUMMARY OF PHASE I RESULTS.....	3
III. DISCUSSION OF PHASE II RESULTS.....	7
III.1 Thermodynamic Cycle Design.....	7
III.1.1 Low Temperature Thermodynamics.....	7
III.1.2 Cycle Design Code.....	9
III.1.3 Tailored Waveshapes.....	11
III.1.4 Diaphragm Working Volumes.....	14
III.2 Regenerator Design and Fabrication.....	15
III.2.1 Requirements.....	15
III.2.2 Flow Distribution.....	15
III.2.3 Design.....	19
III.2.4 Material and Fabrication Techniques.....	22
III.3 Cryocooler Mechanical Design and Fabrication.....	23
III.3.1 Diaphragm Design.....	23
III.3.2 4 K Cooler Layout.....	24
III.3.3 Facility Integration.....	26
III.3.4 Drive Motors and Controls.....	28
III.3.5 Diaphragm Fabrication.....	29
IV. CONCLUSIONS AND RECOMMENDATIONS.....	31
V. ACKNOWLEDGEMENTS.....	31
BIBLIOGRAPHY.....	31

LIST OF FIGURES

1.	180° DUAL STIRLING COOLER WITH RECUPERATIVE REGENERATION.....	1
2.	COUNTERFLOW REGENERATOR TEST APPARATUS SCHEMATIC.....	3
3.	REGENERATOR TEST SECTION.....	4
4.	MEASURED REGENERATOR LOSSES.....	5
5.	HELIUM PROPERTIES.....	7
6.	HELIUM PROPERTIES.....	9
7.	STIRLING REGENERATOR THERMAL LOADING WITH SINUSOIDAL VOLUME CYCLES.....	12
8.	180° STIRLING REGENERATOR THERMAL LOADING WITH SINUSOIDAL VOLUME CYCLES.....	12
9.	TAILORED EXPANDER/SINUSOIDAL COMPRESSOR PRESSURE AND REGENERATOR MASS FLOW WAVES.....	13
10.	EXPANDER PV DIAGRAMS.....	14
11.	COMPUTED FLOW FIELD FOR SINGLE PORT CHANNEL DESIGN.....	16
12.	COMPUTED FLOW FIELD FOR "4 PORT" CHANNEL DESIGN.....	17
13.	"4 PORT" CHANNEL NORMALIZED FLOW VS. LENGTH.....	17
14.	"4 PORT" CHANNEL DESIGN.....	18
15.	HEAT EXCHANGER EFFECTIVENESS VS FLOW DISTRIBUTION UNIFORMITY.....	18
16.	REGENERATOR COMPONENTS.....	20
17.	IMPACT ON MALDISTRIBUTED HEAT EXCHANGER EFFECTIVENESS OF INTERMEDIATE HEADER SECTIONS.....	20
18.	REGENERATOR DETAIL SKETCH (Not to Scale).....	21
19.	REGENERATOR DETAIL (Approx. Scale).....	21
20.	REGENERATOR SECTION, PLATE SPACING.....	24
21.	200 mW 4-20 K COOLER ASSEMBLY.....	25
22.	CRYOCOOLER COLD END ASSEMBLY.....	25
23.	CRYOCOOLER DRIVE SYSTEM COMPONENTS.....	26
24.	4 K COOLER ASSEMBLY.....	27
25.	HEAT SINK HEAT EXCHANGER.....	28
26.	SEMI-TOROIDAL DIAPHRAGM ASSEMBLY.....	30

I. INTRODUCTION

I.1 4 K Cooler Concept

The 4 K cooler operates approximately on the Stirling thermodynamic cycle and embodies an innovation first described by Daney [1] and Roubeau [2]. This innovation embodies two identical Stirling cycle coolers operating 180° out-of-phase and sharing a counterflow heat exchanger in lieu of individual regenerators, as illustrated in Figure 1. This scheme allows heat released by the downward flowing stream to be absorbed by the upward flowing stream, effectively substituting direct gas-to-gas recuperation for the more conventional gas-to-solid matrix regeneration.

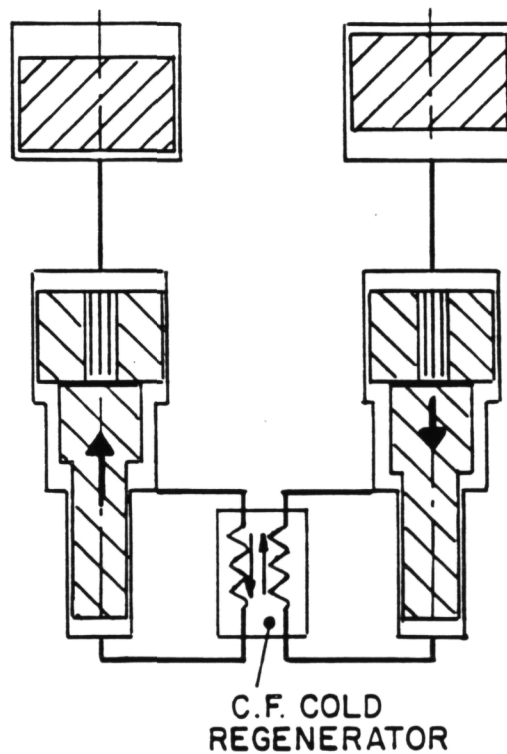


Figure 1. 180° DUAL STIRLING COOLER WITH RECUPERATIVE REGENERATION

The 4 K cooler concept embodies an innovation which perfects the realization of recuperative regeneration. The expander volume is cycled on a non-sinusoidal schedule in order to produce a symmetric waveform of recuperator thermal loading. This innovation is discussed in detail in Section III.1.3.

I.2 Phase I Objective

The objective of the Phase I program was to experimentally demonstrate whether a counterflow regenerator in a multi-module Stirling cryocooler system could provide sufficient and thermally accessible volumetric heat capacity to enable efficient cooling between 4 and 15 K.

I.3 Phase II Objective

The overall technical objective of the Phase II research was to develop and test an innovative Stirling cycle cryocooler with counterflow regeneration to demonstrate cooling capability at 4 K. A nominal 100 milliwatt cooling capacity was selected with heat rejected at approximately 20 K.

A supporting technical objective was to develop fabrication techniques necessary to build a parallel plate counterflow regenerator with:

- plate-to-plate spacing of 37.5 μm ,
- $\pm 1 \mu\text{m}$ channel-to-channel spacing tolerance at all elevations, and
- helium leak tightness of the assembly to 10^{-9} standard cc/sec.

An additional supporting objective was to develop a drive motor electronic control system to provide non-sinusoidal displacer motion required for perfectly balanced counterflow regeneration with two streams. Tailoring the displacer motion waveform as a means of modifying the regenerator thermal loading is, to the best of our knowledge, an original concept with important performance benefits.

The overall technical objective defined a goal which, when met, will advance the state-of-the-art in low temperature closed cycle mechanical cryocoolers.

II. SUMMARY OF PHASE I RESULTS

The specific technical objective of the Phase I program was to experimentally demonstrate whether a counterflow regenerator in a multi-module Stirling cryocooler system could provide sufficient and thermally accessible volumetric heat capacity to enable efficient cooling between 4 - 15 K.

In pursuing this objective, we sought to provide a direct experimental comparison of counterflow and conventional configuration regenerator performance in a single test apparatus capable of being operated in either mode. The experiment is depicted schematically in Figure 2. A simple constant cross-section counterflow regenerator was designed, built and instrumented. The test article is shown to scale in Figure 3. A pair of piston compressors, capable of operating at phase angles of zero or 180°, provided periodic flow through inner and outer flow channels in the regenerator. The flow rate of helium boil off cooling the cold end of the test regenerator was measured in a calorimeter to determine losses.

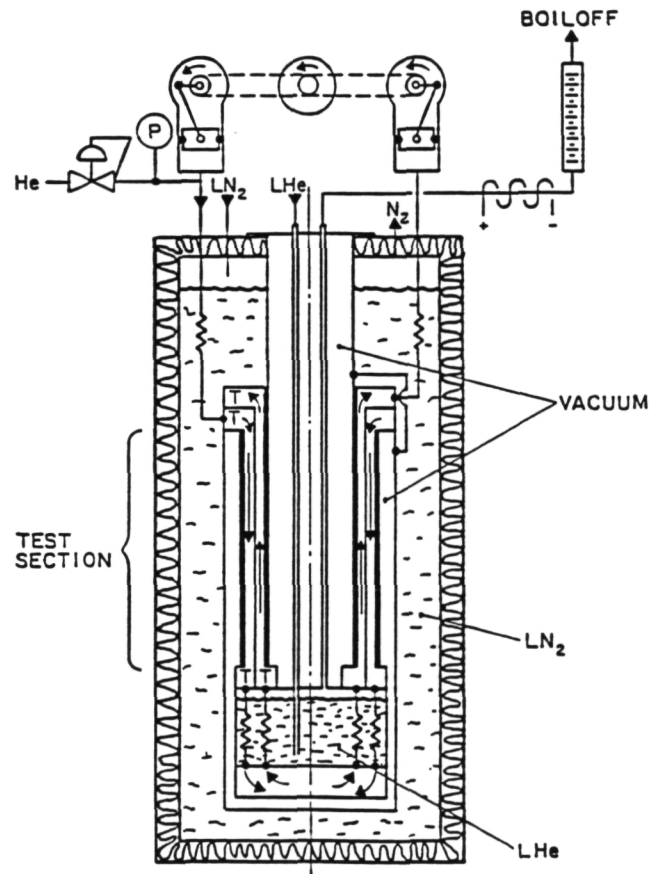


Figure 2. COUNTERFLOW REGENERATOR TEST APPARATUS SCHEMATIC

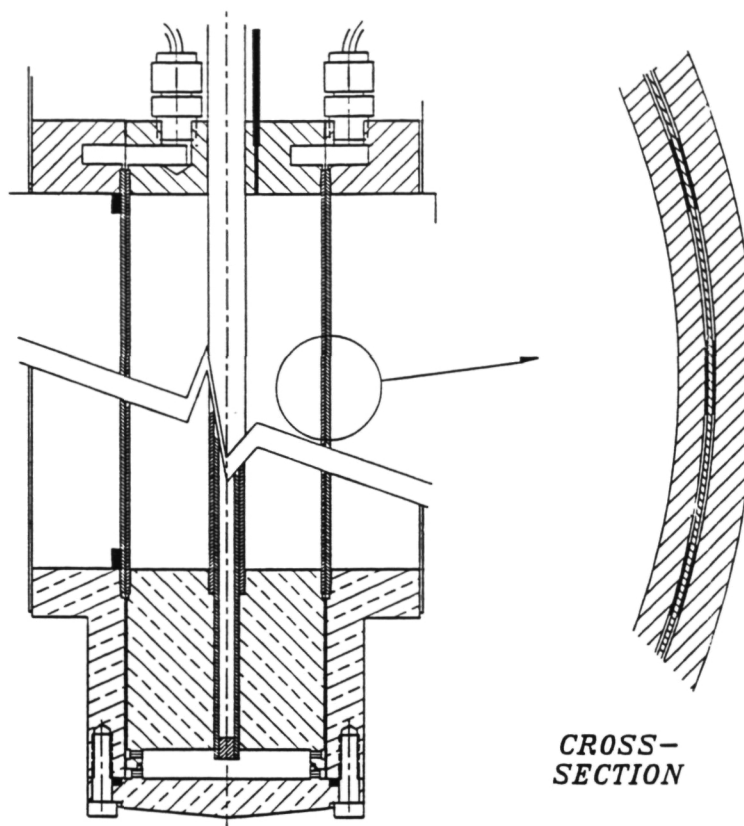


Figure 3. REGENERATOR TEST SECTION

Loss measurements were taken in the standby, counterflow, and co-flow modes, with the latter terminated by freeze up of the apparatus. Figure 4 displays the measured loss rate at standby, with counterflow, and during the early minutes of the startup transient for co-flow, or conventional operation. The standby loss includes both conductive heat leak down the regenerator between 78 K and 4 K and the environmental heat leak of the dewar. As shown by the dotted line of Figure 4, the standby losses were very steady for extended periods at 0.2 watts. The data labeled "counterflow" on Figure 4 show total measured loss over a 20 minute time interval at 2 atm and 10 Hz frequency, with data points taken every minute. The one blip at 12 minutes on the plot is the resolution of the digital calorimeter readout, derived from four individual readings of temperature, voltage and current. Enthalpy flux due to ineffectiveness in the counterflow case is seen to result in a loss of 0.4 watts above the standby loss.

When shifted to the co-flow or conventional mode, the apparatus manifested an immediate clear upward trend in loss, but leveled off at approximately 10 minutes and consequently began a long decline and ultimately approached standby loss levels. Diagnostic tests revealed that impurities in the laboratory grade helium working fluid had frozen out in the cold end of the test section, ultimately completely blocking off flow through the channels. The rising trend of the data represents the thermal time constant of the calorimeter in responding to the step change in boil off following the mode shift from counterflow to conventional. We believe that the change in slope of the data approximately 5 minutes into

the test indicates that freeze up is beginning and the diminishing mass flow is rapidly reducing the losses associated with both heat capacity and heat transfer losses.

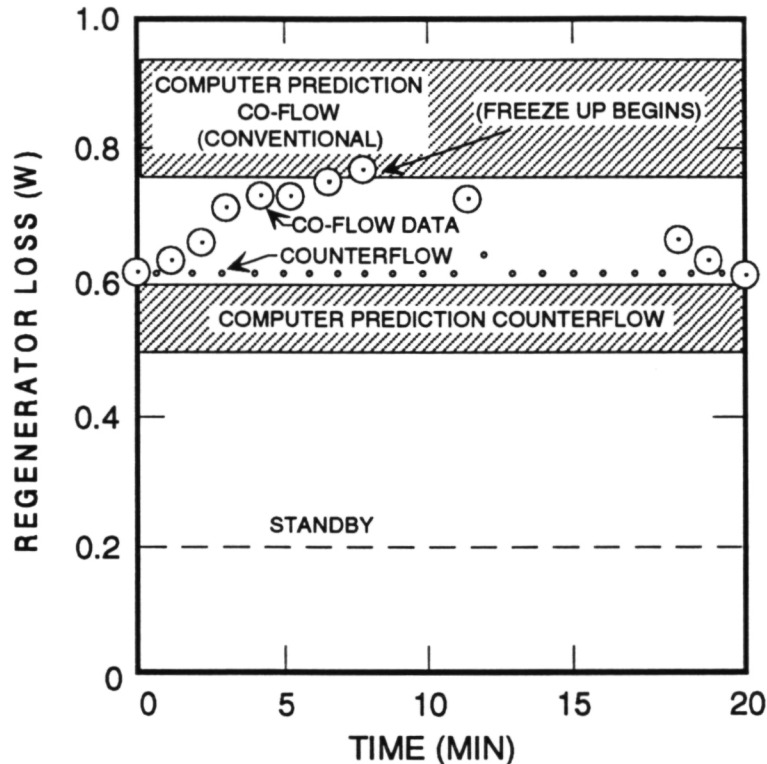


Figure 4. MEASURED REGENERATOR LOSSES

Also included on Figure 4 are loss levels predicted by computer codes developed in house for the design of Stirling cycle cryocoolers. RGRDISC1 (real gas regenerator, discretized) performs mass, momentum and energy balances on regenerator control volumes throughout the operating cycle in order to predict the enthalpy flux loss due to heat transfer and heat capacity effects. In the counterflow mode, this code ignores enthalpy flux losses due to heat capacity effects, effectively assuming infinite matrix heat capacity. This code utilizes real gas thermodynamic properties provided from the Oxford University code HEPROP, and takes input regarding geometry, operation, and temperature profile. For the temperature profile in our calculations, we input the profile measured by means of silicon diodes on the test apparatus. The predicted losses for counterflow operation are displayed on Figure 4. Losses for the co-flow mode are calculated by adding to the mean film ΔT predicted for counterflow, an estimate for the combined gas-wall temperature swing due to finite heat capacity. This

estimate gives a co-flow loss that is 0.2 to 0.4 watts higher than the predicted counterflow loss. The experimental data and analytical predictions are consistent and indicate that counterflow regeneration finesses the dependence of regenerator performance on matrix heat capacity.

Further testing was performed at internal expense with high purity helium at slightly sub atmospheric pressure and leak buffered external plumbing to minimize the potential for recurrence of contamination and freeze up. Unfortunately, the reduced pressure corresponded to reduced mass flow rate, and losses scale as mass flow squared. The resulting loss of signal-to-noise ratio of the experiment rendered the resulting data inconclusive. The initial data, though flawed as a result of experimental difficulties encountered at 4.2 K, provide convincing evidence that counterflow regeneration is technically feasible and should be developed further. A complete low temperature refrigerator conceptual design was therefore developed around the Phase I regenerator concept, as a technology demonstrator for 4 K closed cycle mechanical refrigeration.

III. DISCUSSION OF PHASE II RESULTS

III.1 Thermodynamic Cycle Design

III.1.1 Low Temperature Thermodynamics

Regenerative cooling cycles, such as the Stirling, which operate below the critical temperature of the working fluid (helium) benefit strongly from low operating pressures in three distinct ways. First, the expander provides more gross cooling power at subcritical pressures than at supercritical pressures for the same temperature, swept volume, and cycle pressure fluctuation. Equation 1 below is the standard expression for computing the heat absorbed by the working fluid in an expansion process:

$$dQ = dH - V_E dp \quad (1)$$

Substituting from Maxwell's relations, Eqn. 1 can be manipulated into the form:

$$dQ = \beta T_E V_E dP \quad (2)$$

where β is the isobaric expansivity, a thermodynamic property. In the ideal gas regime, $\beta T = 1$, and Eqn. 2 reduces to the familiar form for computing expansion work in an ideal gas. At temperatures below 10 - 20 K however, helium is most definitely not an ideal gas at supercritical pressures, as illustrated in Figure 5. βT for supercritical pressures rapidly approaches zero as temperature drops below 4 K, driving the gross cooling output of a given expander toward zero. Figure 5 amply illustrates that suitably low pressures enable the expander to retain cooling performance consistent with (or slightly better than) the ideal gas case. This is consistent with the general tendency of very low pressure superheated gases to approximate ideal gas behavior.

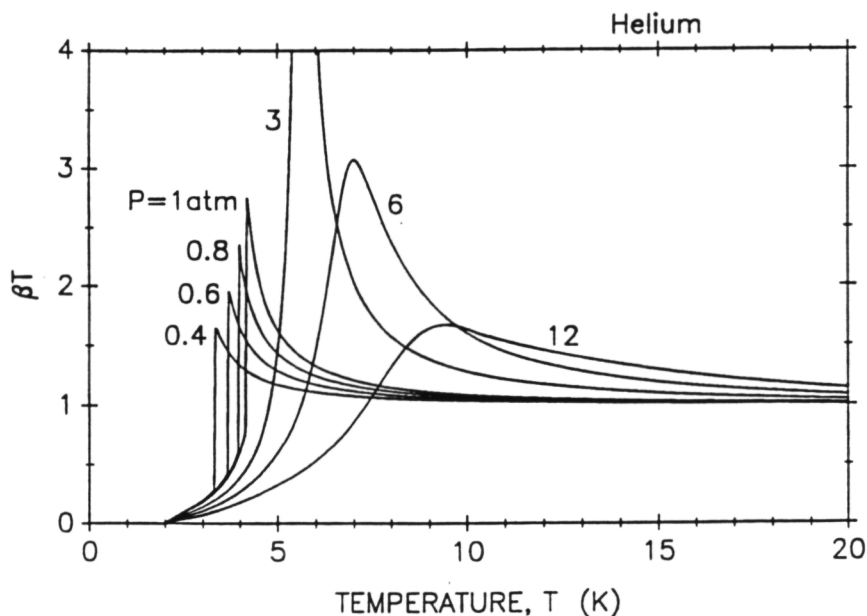


Figure 5. HELIUM PROPERTIES

The second advantage of low pressure operation is rooted in the impact of mass flow rate upon regenerator losses, specifically those associated with cyclic heat transfer between the gas and the regenerator walls. Consider for simplicity's sake only the heat transfer occurring due to the physical motion of a fluid element up or down the regenerator temperature gradient, eg. the regenerative heat transfer. Equation 3 describes the instantaneous regenerative heat flux at some arbitrary location in the regenerator.

$$\dot{Q}_R = \frac{\dot{m} C_p}{A_{HT}} \frac{dT}{dx} \quad (3)$$

Note that this formulation ignores the additional heat transfer burden resulting from simultaneous compression or expansion of the fluid element due to the cycling system pressure. The difference between wall and bulk gas temperature, ΔT_{film} , must be adequate to drive the instantaneous heat flux computed by Equation 4, given the local heat transfer coefficient:

$$\dot{Q}_R = h_{film} (T_w - T_{BULK}) \quad (4)$$

Including a suitable sign convention for the direction of heat transfer in Eqn. 4 demonstrates that gas temperature on the downward "blow" is slightly above local wall temperature, and gas temperature on the upward "blow" is slightly lower than local wall temperature. The manifestation of cyclic heat transfer loss is thus a net fluid enthalpy flux down the regenerator channel. The loss rate arising from this enthalpy flux is calculated as:

$$\dot{Q}_{HT LOSS} = \dot{m} C_p (T_w - T_{BULK}) \quad (5)$$

The cyclic integral of Eqn. 5 defines the energy which travels down the regenerator to the expander in one cycle of operation due to heat transfer loss. Inspection of Eqns. 3, 4 and 5 reveals that heat transfer loss varies as the square of mass flow rate. Heat transfer loss subtracts directly from gross cooling power and thereby diminishes net cooling power, so regenerator mass flow rate should be minimized. This line of reasoning indicates that gross expander cooling power per unit of mass flow rate provides a figure of merit for maximizing cycle efficiency.

$$\frac{\dot{Q}_E}{\dot{m}} = \frac{f \beta T V_E \Delta P}{f V_E \rho} \quad (6)$$

Equation 6 shows by inspection that net performance is enhanced by reducing system pressure (density).

The third and final advantage of low pressure operation is its tendency to eliminate peaks in important thermodynamic properties such as specific heat capacity, as illustrated in Figure 6a. Physically, a heat capacity peak means that more heat transfer per unit of gas temperature change will be necessary in one portion of the regenerator than in others as

gas is displaced through it. This in turn tends to flatten the temperature gradient in that section of the regenerator, making prediction of the temperature profile difficult and its stability questionable. Gas and metal properties are both quite temperature dependent at very low temperatures, and correct design of the regenerator requires an accurate knowledge of the temperature distribution. As can be seen in Figure 6b, low pressure operation avoids the problems of non-ideal variation in the helium gas properties.

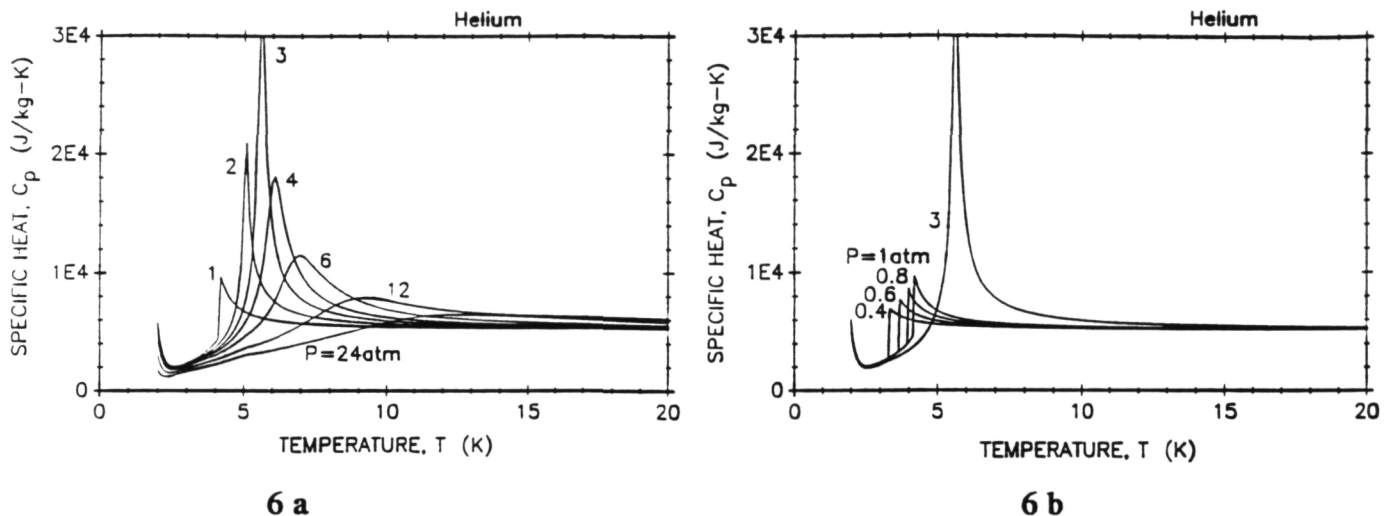


Figure 6. HELIUM PROPERTIES

Based on the preceding discussion, a maximum cycle pressure of 0.6 atm was baselined for the thermodynamic cycle design. Very low cycle pressures are beneficial, and perhaps necessary, for Stirling cycles lifting from 4 K. A low pressure multi-stage cycle lifting from 4 K all the way to 300 K however would be unacceptably large and heavy at the warm end. Cycle optimization studies demonstrated clear advantages in size and weight with a hybrid cycle approach, cascading a multi-stage high pressure Stirling (or other) cycle cryocooler above a low pressure Stirling cycle operating between 4 K and approximately 20 K. The thermodynamic design pursued in this program was therefore for a single stage low pressure Stirling cycle lifting to 20 K.

III.1.2 Cycle Design Code

The thermodynamic cycle was designed with the assistance of a proprietary computer code, FORTIT B, developed under previous SBIR projects. FORTIT is a simple numerical code which starts in the isothermal expander with a prescribed pressure and volume waveform and marches up through the regenerator to the compressor, computing heat transfer and pressure drop parameters from laminar, quasi steady relationships. Outputs include the compressor swept volume and volume waveform necessary to have provided the originally assumed pressure waveform. Gas in the regenerator is initially assumed to be in local thermal equilibrium with the regenerator walls, and the temperature and pressure is used to compute the thermodynamic state in each regenerator node. Change in state from node to node is then used to compute instantaneous heat flux, from which the local film ΔT is computed by means

of a quasi steady laminar Nusselt number. The actual gas temperature in each node thus computed is then used to call up the gas enthalpy from the Oxford University helium properties code HEPROP. The product of enthalpy and mass flow rate provides the instantaneous enthalpy flux up or down the regenerator in each node. Cyclic integration of the enthalpy flux provides the net enthalpy flux down the regenerator in each node, the loss associated with heat transfer irreversibility.

FORTIT also calculates the conduction of heat down the metal of the regenerator, using the appropriate temperature dependent metal conductivity from a database. Both conduction and enthalpy flux depend upon the local temperature gradient in the regenerator. For steady state conditions, the First Law requires that the sum of conduction and enthalpy flux is constant for all regenerator nodes. Starting from a pure conduction temperature profile, the code iterates on temperature profile until the First Law criterion is satisfied within prescribed convergence limits. The code acronym derives from FORtran Temperature ITeration.

The designer inputs variations on the key operating and geometric parameters such as frequency, minimum pressure, pressure ratio, regenerator length, width, channel height, and wall thickness. Through a manual iteration process, the designer explores the computed impact of each design parameter change to converge on a well balanced design. To assist the designer in parametric exploration, the code outputs system level data such as efficiency, gross cooling, and net cooling as well as nodal details of regenerator performance such as heat flux, film ΔT , viscous pressure gradient, etc. The system level outputs show the designer what works well, and the nodal outputs provide insight as to why. Each code run requires approximately 30 seconds on a PC, allowing an experienced designer to converge on a reasonably well balanced cycle design in a few days or less. Pre and post processor and graphics routines have been wrapped around the code to facilitate pre-programmed parametric sensitivity studies in an automated mode. No attempt has been made to wrap around an optimization routine, as we believe that the designer's understanding of system level non-thermodynamic design issues is critical to developing thermodynamic designs which are practical to implement.

Manual iteration with the design code resulted in convergence on the geometric and operating design parameters displayed in Table 1.

Table 1. CRYOCOOLER DESIGN PARAMETERS

Load Temperature	4 K
Load	200 milliwatts
Sink Temperature	20 K
Expander Volume (2)	0.75 cubic centimeters each
Pressure Ratio	2.0
Minimum Pressure	0.03 MPa
Compressor Volume (2)	5.84 cubic centimeters each
Operating Frequency	10 Hz
Regenerator Length	5 centimeters
Regenerator Width	75 centimeters
Channel Height	37.5 μm
Wall Thickness (304 ss)	37.5 μm

III.1.3 Tailored Waveshapes

Recuperation overcomes the low temperature performance deterioration of conventional regenerator matrix materials due to declining heat capacity. Effective implementation of the recuperative Stirling concept at low temperatures however depends for its success on further thermodynamic innovations. First, the instantaneous rate of heat transfer out of one gas stream must match with high precision the instantaneous rate of heat transfer into the other gas stream throughout the thermodynamic cycle, requiring temporal symmetry in the recuperator thermal load waveform. Mechanical implementation of the Stirling cycle typically invokes sinusoidal waveforms of compressor and expander volume in time for reasons of kinematic simplicity and/or resonant operation. Such machines however produce a decidedly asymmetric cycle of mass flow (and thermal loading) on the regenerator, as illustrated in Figure 7. This thermal loading asymmetry results in imperfect recuperative cancellation in the 180° dual Stirling cycle implemented with sinusoidal volume waveforms, as shown in Figure 8. The uncanceled residual thermal load in Figure 8 represents a net regenerative demand on the heat capacity of the metal in the heat exchanger. The present 4 K cooler concept perfects the implementation of the recuperative Stirling invention by distorting or "tailoring" the expander volume waveform so as to provide symmetric thermal loading waveforms and perfect recuperation. This result can be achieved for waveform tailoring of the expander, the compressor, or both, and an infinite number of symmetric thermal load combinations are possible. The volume waves which produce symmetric thermal load waves, however also influence the expander PV diagram (gross cooling) and the heat exchanger effectiveness loss. Optimum volume cycles will provide recuperative symmetry while also maximizing net expander cooling output. The design reported herein used a sinusoidal compressor volume and skewed sinusoid expander volume cycles to provide recuperative symmetry, maximized net cooling, and relatively simple drive and control requirements. Figure 9 illustrates the selected expander volume waveform and the resulting waveforms for system pressure and regenerator mass flow rate (which corresponds relatively well to thermal load). Figure 10 compares the expander PV for the Figure 9 cycle with a sinusoidal cycle with similar net cooling output.

The second important concept in implementing any successful low temperature thermodynamic cycle is to harmonize the important cycle parameters with the real gas thermodynamic properties of the helium working fluid. Section III.1.1 provided a detailed discussion in this area and explains why the 4 K cooler concept employs extremely low working gas pressures (0.6 atm max.).

The final key thermodynamic innovation in the 4 K cooler concept is the use of elastic metal diaphragms to form the positive displacement compressor and expander volumes. The diaphragms define, seal, and displace the working volumes. This geometry is chosen primarily because it provides expansion and compression processes very close to the ideal isothermal processes which grace the Stirling cycle with Carnot efficiency in the ideal limit. Practically speaking, diaphragms also serve the function of reciprocating bearings and seals, providing a non-wearing mechanism with high reliability and long life potential. Thermodynamic aspects of diaphragm working volumes are discussed in Section III.1.5 and the mechanical aspects in Section III.3.1.

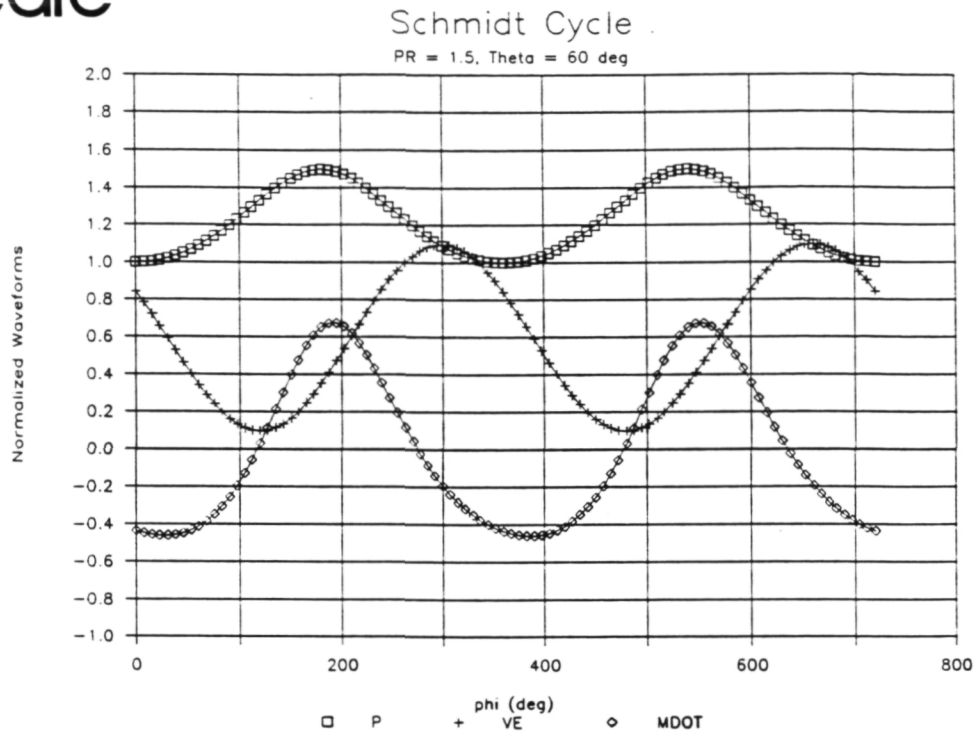


Figure 7. STIRLING REGENERATOR THERMAL LOADING WITH SINUSOIDAL VOLUME CYCLES

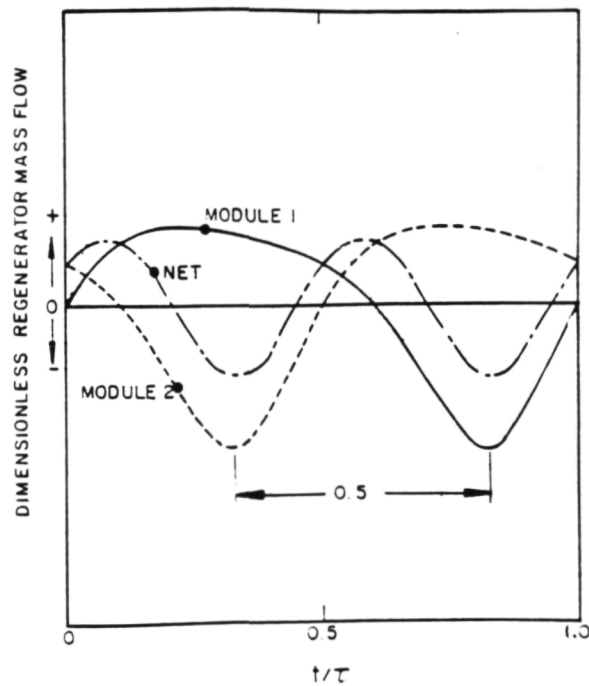
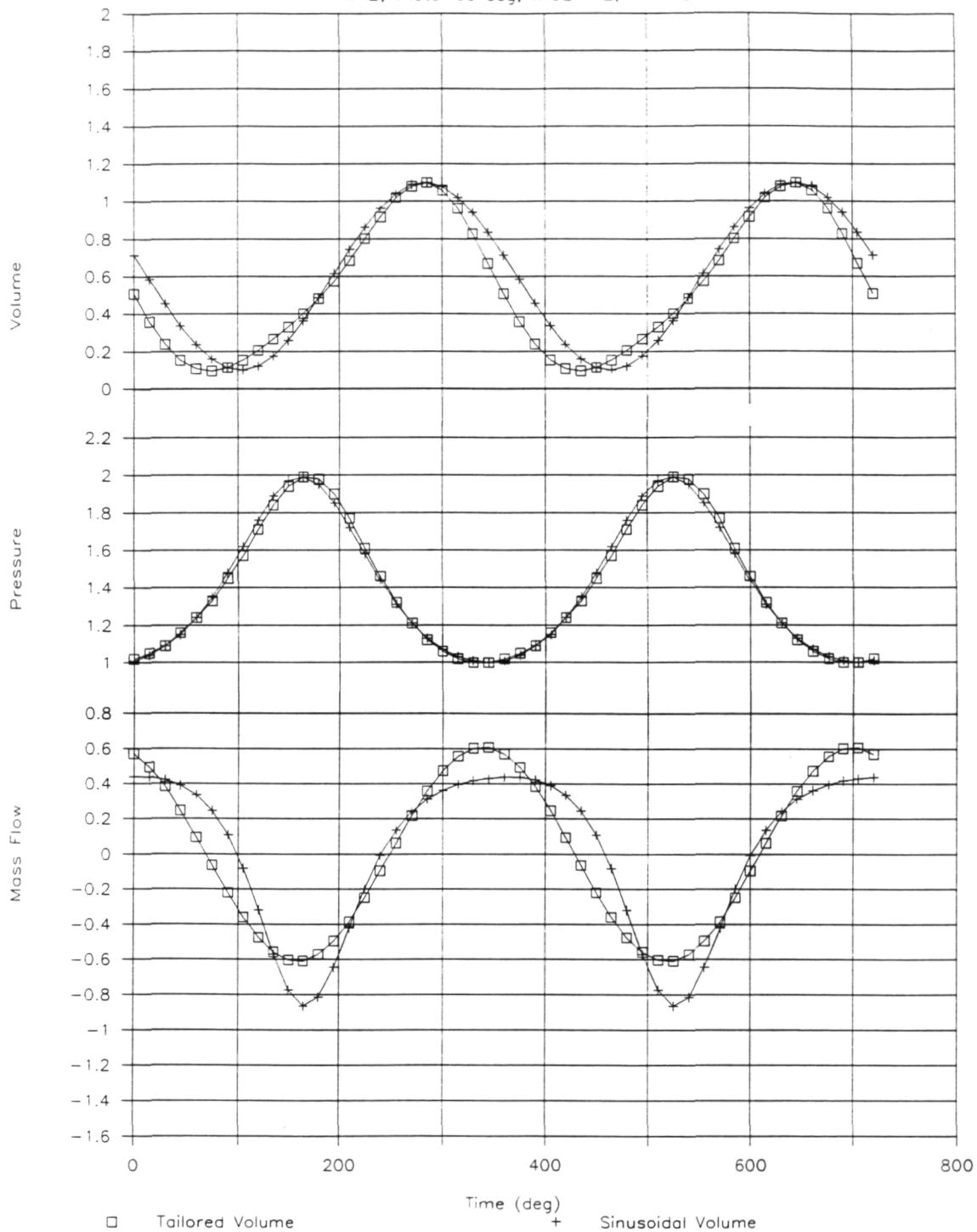


Figure 8. 180° STIRLING REGENERATOR THERMAL LOADING WITH SINUSOIDAL VOLUME CYCLES

Sine Wave MDOT and VP

PR=2, Theta=60 deg, TAUE=4.2, XR=4.9



**Figure 9. TAILORED EXPANDER / SINUSOIDAL COMPRESSOR
PRESSURE AND REGENERATOR MASS FLOW WAVES**

Sine Wave MDOT and VP

PR=2, Theta=60 deg, TAU=4.2, XR=4.9

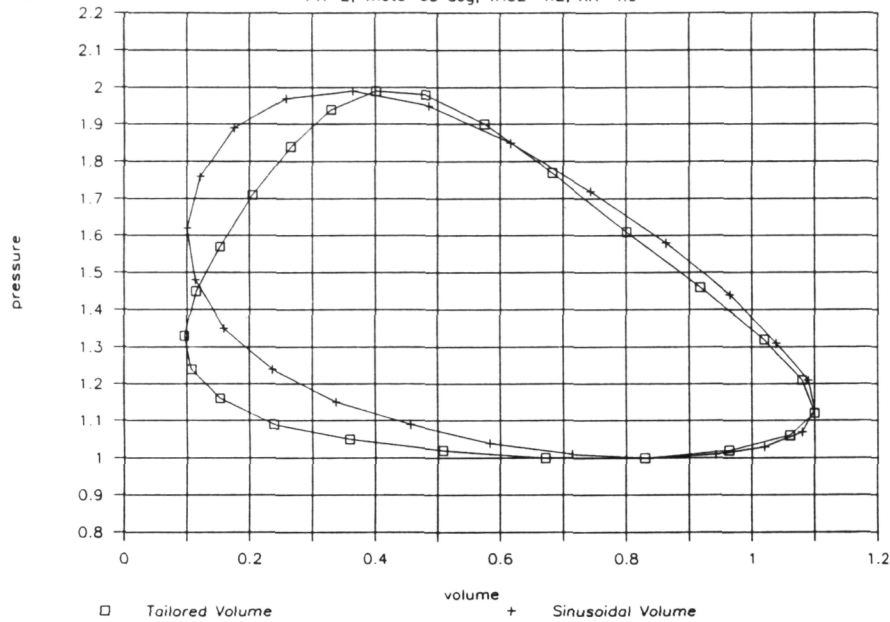


Figure 10. EXPANDER PV DIAGRAMS

III.1.4 Diaphragm Working Volumes

The expander diaphragms are approximately 10 centimeters in diameter and have a total stroke of approximately 0.2 millimeters. Design of the diaphragms must satisfy a number of criteria. First, the requisite swept volumes specified in the thermodynamic cycle design must be provided. Second, the combination of stroke, diameter, and thickness must result in peak stresses well below the fatigue endurance limit, considering both deflection and pressure loading stresses. Finally, the stroke must be kept small enough that compression and expansion heat transfer processes do not depart significantly from isothermal.

Structural design of the diaphragms to maintain suitably low stresses is described in Section III.3.1. For the 4 K expander, the near isothermal criterion turns out to be most limiting, resulting in extremely low stress levels and long operating life expectations with no hazard of fatigue failure.

To compute process departure from isothermality, we calculated instantaneous isothermal heat flux from Eqn [2] from Section III.1.1 throughout the expander cycle. Eqn [4] from the same section is then used to calculate instantaneous bulk gas temperature, conservatively assuming purely conductive heat transfer through stagnant gas at the appropriate instantaneous value of expander stroke (gas volume thickness). Design stroke is selected such that the computed bulk gas temperature transient is < 10% of the adiabatic temperature rise.

Finally, we assessed the impact of pressure drop from radial flow of gas in and out of the cycling expander volume. A separate effects code, XPANDIT, was written for this purpose. Based upon these laminar flow calculations, we conclude that a minimum clearance gap of 1.5 mils limits radial pressure drop to an insignificant fraction of the cycle pressure swing.

Based on the calculations described above, we specify the expander stroke and diameter at 0.368 mm and 6.35 cm, respectively.

III.2 Regenerator Design and Fabrication

III.2.1 Requirements

The 4 K cooler regenerator must provide the length, width, and channel spacing called out in Table 1 of Section III.1.2. It must provide adequate precision in plate-to-plate spacing to ensure uniform flow distribution, as discussed further in the following section. The individual regenerator plates are subjected to reversing cycles of ΔP and must be adequately stiff and supported to prevent resultant strains which are a significant fraction of nominal plate-to-plate spacing.

The regenerator forms part of the system pressure boundary and must provide hermetic leak tight integrity to ensure retention of the working gas over extended time periods. The regenerator must also endure an arbitrary number of cool down/warm up cycles without incurring undue thermal stresses or joint failures. The perimeter joinery details must not introduce additional conductive heat leak beyond a small fraction of that associated with the indispensable plates and accounted for in design calculations.

Finally, the manifolding at the ends of the regenerator must compromise between minimizing dead volume while holding dynamic head to $< 1/10$ of the viscous pressure drop in the regenerator flow channels.

III.2.2 Flow Distribution

Regenerator flow maldistribution, either across the individual channel width or amongst parallel channels, can erode the net performance of the heat exchanger and reduce cooling capacity and efficiency of the cryocooler. For this reason, thermodynamic design of the 4 K cooler in this program incorporated a margin of two on cooling capacity. One hundred milliwatts of cooling at 4 K had been suggested as a reasonable and useful performance goal in formulating the program objectives. A 100% margin was considered tactically prudent, given the unproven nature of the 4 K cooler innovations. The thermodynamic cycle design presented in Table 1 of Section III.1.2 incorporates this 100% design margin primarily to anticipate the erosive nature of regenerator performance shortfalls, specifically from flow maldistribution at extremely low gas viscosities in a low pressure drop regenerator design.

The commercial CFD code, FLUENT, was employed to compute the lateral distribution of flow across the regenerator channels as a function of the number of inlet and discharge ports. The simplest geometry, with a single inlet and discharge port at diagonally opposite corners of the regenerator produced the computed flow field illustrated in Figure 11. Significant distortions of the flow field were predicted over 25% of the regenerator length at each end with this geometry. While it is obvious that a very fine mesh of small ports is desirable from a flow distribution sense, the separation between ports includes an irreducible minimum gasket width for reliable sealing. Increases in the number of ports therefore reduce the fractional plate width available for flow porting, in order to accommodate the increased number of gaskets.

A design was finally selected which effectively divides the channel into four symmetric parallel flow streams with one inlet and one discharge port each. Because the flow systems are symmetric about the port center lines, this translates into two full ports at one end and one full port flanked by two half ports at the other end. This design confines significant lateral flow distortions to approximately 10% of the regenerator length at each end. The FLUENT output is shown in Figure 12, and the imbedded distortion data are amplified in Figure 12. Calculations were performed at several planes across the hydraulic diameter, and Figure 13 displays the bulk integrated flow distortion results. Figure 14 displays the design implementation of the "four port" scheme.

The second concern in regenerator design is non-uniform flow distribution across the many parallel channels of the regenerator caused by the expected tolerance on plate-to-plate spacing. This tolerance would arise both from material thickness tolerance and from variations in material yield during a diffusion bond fabrication process. Although research indicated that plate-to-plate spacing precision of 1 micron tolerance could be held, the extreme sensitivity of channel flow to plate spacing suggested that channel flow variations of 25% could result. This degree of flow imbalance in turn would have a severe impact upon regenerator effectiveness, as illustrated in Figure 15. The parameter C_1/C_h corresponds to the mass flow ratio between adjacent channels, and the parameter F_1 indicates the fraction of the heat exchanger channels afflicted by the mass flow imbalance. Clearly, a 25% flow imbalance over 50% of the regenerator channels could reduce the nominal effectiveness by approximately 7 percentage points. Steps taken to combat this sensitivity are discussed in Section III.2.4 (Design) and their effectiveness in III.2.6 (Inspection).

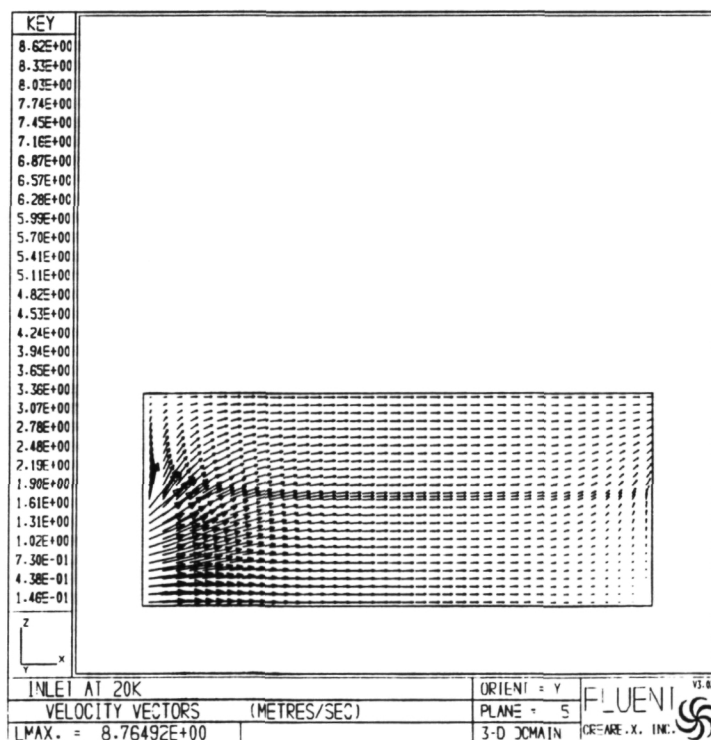


Figure 11. COMPUTED FLOW FIELD FOR SINGLE PORT CHANNEL DESIGN

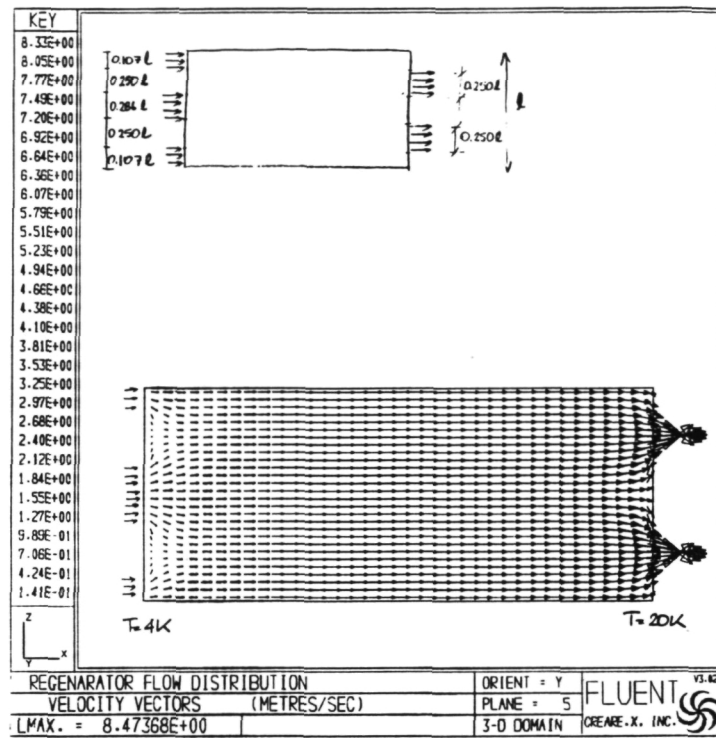


Figure 12. COMPUTED FLOW FIELD FOR "4 PORT" CHANNEL DESIGN
Plot of Position vs. U-Velocity for Geometry G3

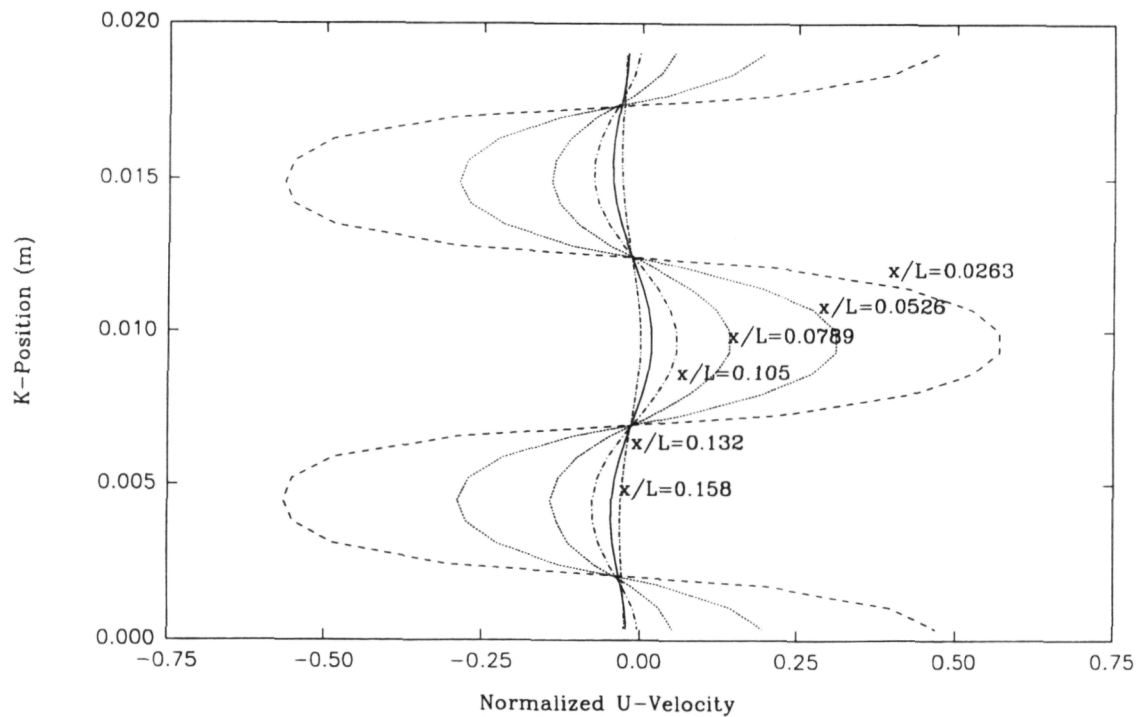


Figure 13. "4 PORT" CHANNEL NORMALIZED FLOW VS. LENGTH



Figure 14. "4 PORT" CHANNEL DESIGN

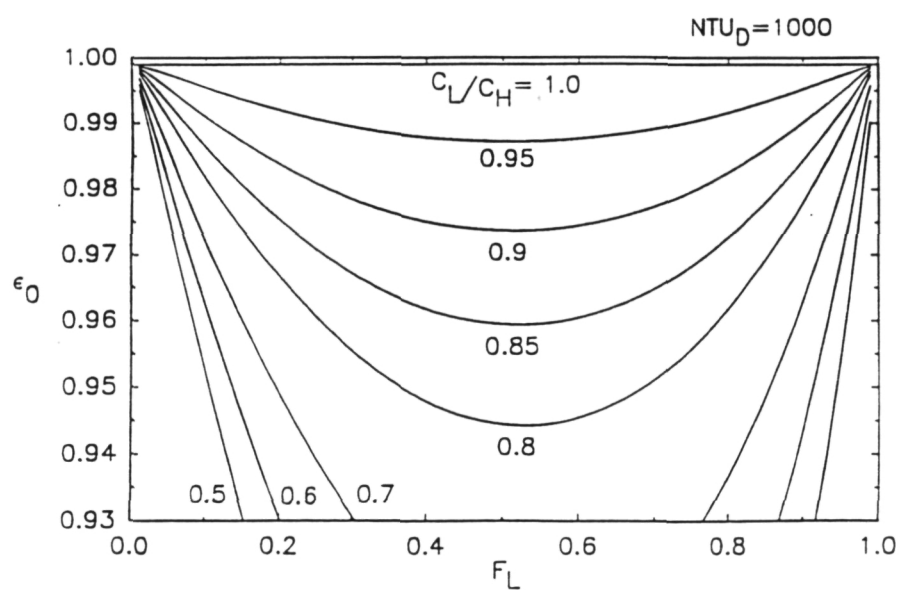


Figure 15. HEAT EXCHANGER EFFECTIVENESS VS. FLOW DISTRIBUTION UNIFORMITY

III.2.3 Design

The regenerator is a parallel plate heat exchanger consisting of 40 "A" channels alternating with 40 "B" channels serving the "A" and "B" Stirling cycles. Porting into each channel consists of two full ports at one end and one full port plus two half ports at the other, as described in Section III.2.2, and illustrated in Figure 16. Plate spacing is maintained by a matrix of spacer buttons which constitute about a 10% flow blockage (included in the thermodynamic design calculations). The pitch between spacers is based upon maintaining acceptably small deflections of the heat exchanger "plates" relative to normal plate spacing under the influence of cyclic channel-to-channel pressure fluctuations. Channel perimeters are sealed by a thin metal "gasket" of thickness similar to the plate spacing buttons.

To attenuate the impact of potential flow maldistribution, two design features are incorporated in the regenerator. The first provides multiple transverse headers between alternating flow channels down the length of the regenerator. Figure 17 from Fleming [3] shows the effectiveness of transverse headers in reducing the performance impact of maldistribution by promoting periodic flow redistribution. The second design feature, which we believe may constitute a patentable invention, is intended to promote transverse thermal mixing between maldistributed flow streams. This invention constitutes transverse copper "bars" which readily conduct heat perpendicular to the flow direction, effectively remixing the gas temperature in all channels at any elevation. In the simple parallel plate heat exchanger configuration, each flow stream exchanges heat only with the two adjacent channels, and performance is very dependent upon local flow matching in those channels. With transverse conductors, the gas in each channel is also able to exchange heat (via the conductors) with gas many flow channels distant on either side. This invention effectively increases the statistical probability of having a good C_l/C_h ratio by thermally connecting a large number of channels subject to a random value distribution of individual channel flow rate and heat load parameters. The invention could find good application in any miniature heat exchanger, where channel tolerance is a large fraction of the nominal channel dimension. Additionally, the invention could be useful for combating the thermal instability problems of cryogenic heat exchangers, where temperature dependent viscosity effects provide positive feedback to exacerbate flow maldistribution.

Implementation of this design concept is illustrated in Figure 18 which represents an edge view section of the concept. Note well that the vertical scale is compressed relative to the horizontal scale in this figure in order to show the pattern of the transverse conductor spacing relative to the "A" and "B" channels. The conductors (copper wires) are much bigger than they appear in this figure. To emphasize this point, Figure 19 is a sketch which is properly scaled in both directions and focusing on a single wire detail. Note that the clearance between the transverse conductor "wire" and the hole through the spacer "donut" provides a cross header through which alternating flow channels can redistribute their flow. This transverse header and conductor detail is terminated at the extreme ends of the plate stack with two blind side plates which capture the transverse conductor wires and complete the sealing of the gas boundary.

Finally, the flow ports at each end of the regenerator, which align in transverse channels, are connected on one side of the plate stack by a manifold block and headered into a single "A" and "B" port which connects to the expanders and compressors by a single round tube each. Ports and headers are sized such that the maximum dynamic head is < 0.10 of the viscous pressure drop down the individual flow channels, in order to promote uniform flow distribution.

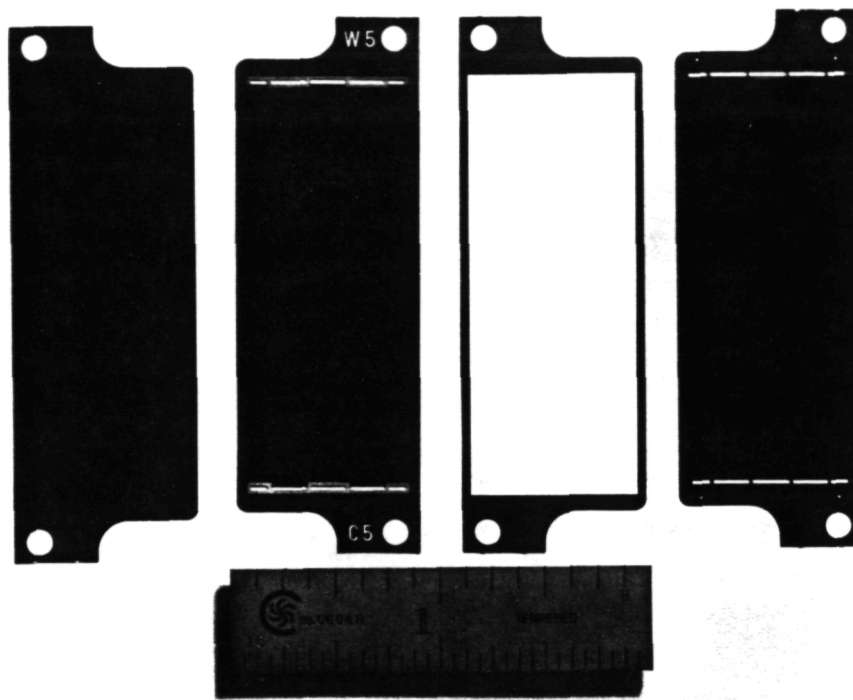


Figure 16. REGENERATOR COMPONENTS

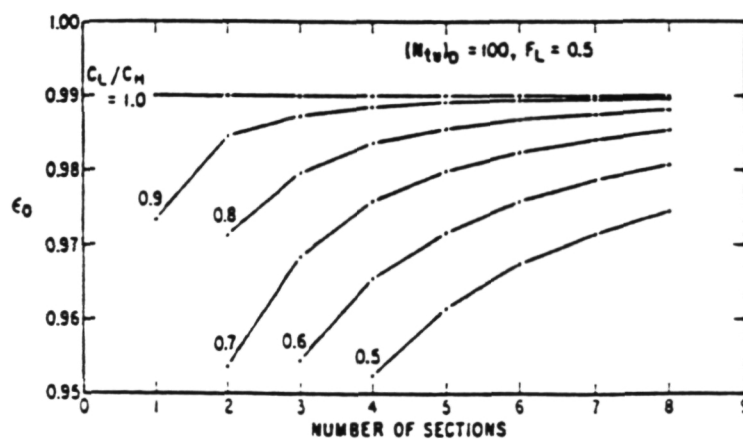
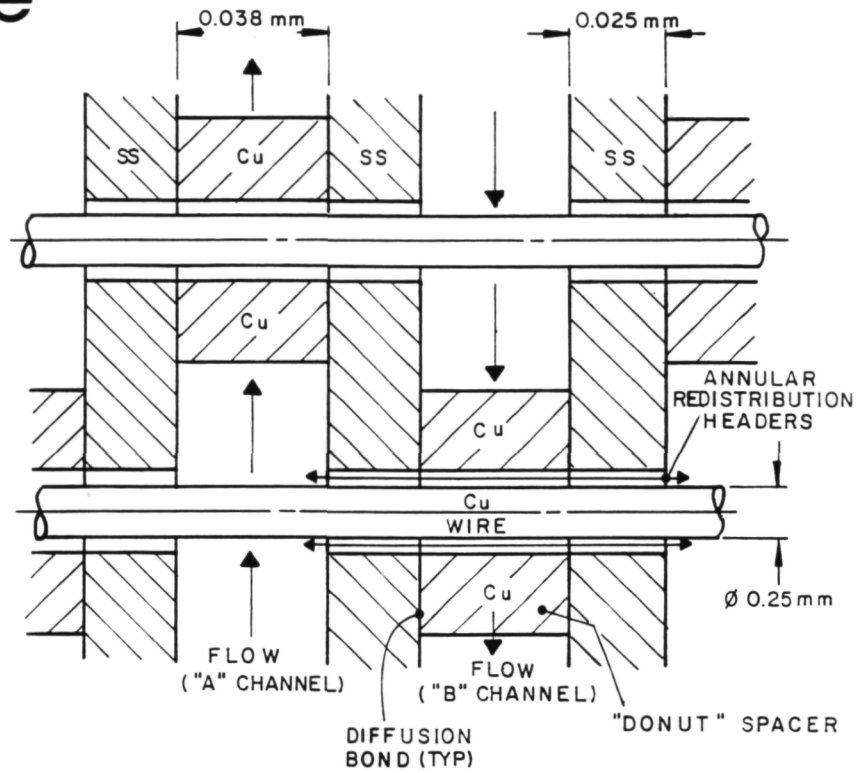


Figure 17. IMPACT ON MALDISTRIBUTED HEAT EXCHANGER EFFECTIVENESS OF INTERMEDIATE HEADER SECTIONS



Figures 18. REGENERATOR DETAIL SKETCH
(Not to Scale)

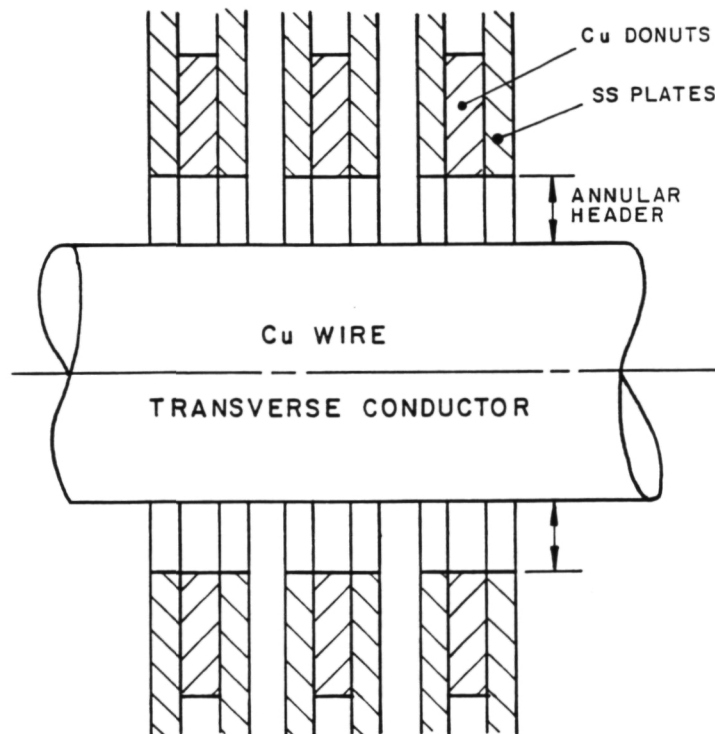


Figure 19. REGENERATOR DETAIL
(Approximate Scale)

III.2.4 Material and Fabrication Techniques

The heat exchanger plates are fabricated of 38 micron (1.5 mil) Type 304 stainless steel. The plate perimeter gaskets are also of 38 micron Type 304 stainless. The pure copper spacer buttons are 42 microns (1.7 mils) thick. The regenerator plates and perimeter gaskets are assembled by diffusion bonding.

Chemical photo etching processes are used to form the regenerator plate details and the perimeter gasket pieces. The etching process includes locating features so that the individual components may be precisely aligned on fixturing pins for accurate registration prior to diffusion bonding.

The individual heat exchanger plates consisted of a 38 micron stainless foil sheet with a finely pitched pattern of 42 micron tall by 100 micron diameter copper spacer buttons and "donuts". The starting material for fabrication of these plates was by metallic foil of stainless and copper. The copper layer was masked and photo etched with a nitric acid solution to remove all but the desired detail. Due to undercutting involved in the photo etching process, the 43 micron tall spacer buttons formed by the process showed base diameters of 125 microns and tip diameters of 75 microns. These spacer buttons would plastically deform in height by 5 microns during diffusion bonding of the final assembly. We first attempted to procure bi-metallic foil formed by diffusion roll bonding of copper sheet onto stainless sheet. Samples of material formed by this process however showed, upon etching away the copper layer, perforation of the stainless sheet induced by the high sheer deformation of the copper layer during roll bonding. We subsequently qualified suitable bi-metallic foil formed by sputtering of copper onto a stainless substrate. The sputtered material is inspected by micrometer measurements for overall thickness and for individual layer thickness by etching away the copper with a 20% nitric acid solution immersion for 90 seconds. The etch completely removed the copper and left the stainless unscathed. Variations in the overall and individual layer thicknesses were approximately ± 1 micron. Microscopic examination of the material showed it to be free of tears or other imperfections.

The bi-metallic foil was subsequently etched to provide the four basic parts from which the regenerator was assembled, shown from left to right in Figure 16 (pg 20): a featureless outside cover plate, a Type A heat transfer plate with stainless substrate and copper features for spacer buttons and manifold gaskets, a stainless rim gasket piece, and a Type B heat transfer plate. A and B plates provide alternate manifold porting for the alternating counterflow channels. The complete regenerator requires two cover plates, approximately 40 each of the Type A and B heat transfer plates, and 80 of the stainless perimeter gaskets. Figure 14 (pg 18) provides a magnified view of one end of a heat transfer plate, showing clearly the copper manifold gasket feature and (less clearly) the copper spacer button features.

Numerous trials were conducted to develop the process parameters for successful diffusion bonding of the stainless to copper and stainless to stainless joints. Stainless to copper joints proved straightforward to make on sample pieces, and joints were successful for both mechanical strength and hermetic integrity. Stainless to stainless joints proved sensitive to a need for aggressive cleaning procedures. Diffusion bonding at high temperature, pressure, and time produced successful stainless to stainless hermetic joints but also resulted in unacceptable plastic deformation of the stainless parts. Control of the net stainless thickness is critical to ensuring correct and uniform plate-to-plate spacing in the assembled regenerator. Trials with reduced levels of pressure and temperature applied over a longer period of time produced minimal distortion with 1800°F process temperature. Samples were leak tested with a helium

mass spectrometer and showed leak tight integrity of the external pressure boundary of 10^{-9} std cc/sec. Stainless sample pieces with a thin sputtered on copper flash were also successfully diffusion bonded at 1500°F. When diffusion bonding, care was taken to ensure that the entire stack came up to bonding temperature before applying compressive load with the hydraulic ram platens, to ensure that all layers of the stack would have completed their lateral thermal expansion before clamping loads were applied.

Multi plate sample stacks of diffusion bonded regenerator components were potted and sectioned for destructive microscopic inspection of internal plate spacing precision. Internal plate-to-plate spacing uniformity was found to be excellent as shown in Figure 20. Periodic flaws were however observed in bonding of the stainless plates to the copper gasket details in the manifold porting regions, a defect which would result in internal leakage between the two sides of the regenerator. In this region, the stainless steel foil is pinched between a continuous copper gasket on one side and three small copper buttons on the other. Under diffusion bonding conditions, the three buttons proved inadequately stiff to press the stainless against the copper gasket firmly enough to ensure a proper sealing bond. A design modification was conceived for future regenerator fabrication to positively address this process problem. The three small copper buttons on the stainless plate in the manifold porting region would be eliminated in the original photo etching pattern. In their place, stainless steel tabs on revised perimeter gasket pieces would project downward across the manifold ports and cover the region previously occupied by the three copper buttons. These tabs would provide continuous support of the port gasket region during diffusion bonding and would be sputtered with silicon dioxide to prevent sticking to the stainless steel plates on either side of the tabs. After diffusion bonding, the support tabs would be cut off by wire EDM and manually removed from the bonded regenerator. All issues pertaining to materials and process development were successfully demonstrated on prototypical regenerator components with the exception of perfecting port gasket sealing in the diffusion bond process. Minor design revisions were developed to positively resolve this remaining detail, but were not implemented due to funding limitations. These development efforts provide confidence that a miniature plate-type heat exchanger with one micron plate-to-plate spacing precision can be successfully fabricated for a 4 K cooler or other applications.

III.3 Cryocooler Mechanical Design and Fabrication

III.3.1 Diaphragm Design

Diaphragms in the 4 K cooler design were of Type 304 stainless steel shim stock. Deflections were on the order of 0.25 mm, and gas pressure differential was a maximum of 0.6 atms. Thickness of the diaphragms was based upon two criteria, keeping the combined stress from deflection and pressure loading beneath 100 MPa and keeping the volumetric bulge due to pressure differential beneath 5% of the swept volume. Application of these criteria resulted in selection of 0.25 mm thickness for expander diaphragms and 0.50 mm thickness for compressor diaphragms.

The diaphragm design incorporates a thick rigid hub and a thin flexible annular web region with radius ratio of about 0.6. The annular geometry in these proportions provides the lowest peak elastic stress per unit of volume swept. The 100 MPa peak stress limit falls well below the 99.999% fatigue endurance probability at 10^{-10} cycles of reversing stress, well beyond 10 years of operating life at the 10 Hz design frequency. Diaphragms therefore provide the basis for a highly reliable long life 4 K cooler with a simple, passive, and robust reciprocating bearing system. The machine design, discussed further in the following section,

incorporates pairs of diaphragms on a common axis linked by a thin walled structural tube, constraining the system to a single degree of freedom along the axis of reciprocation.

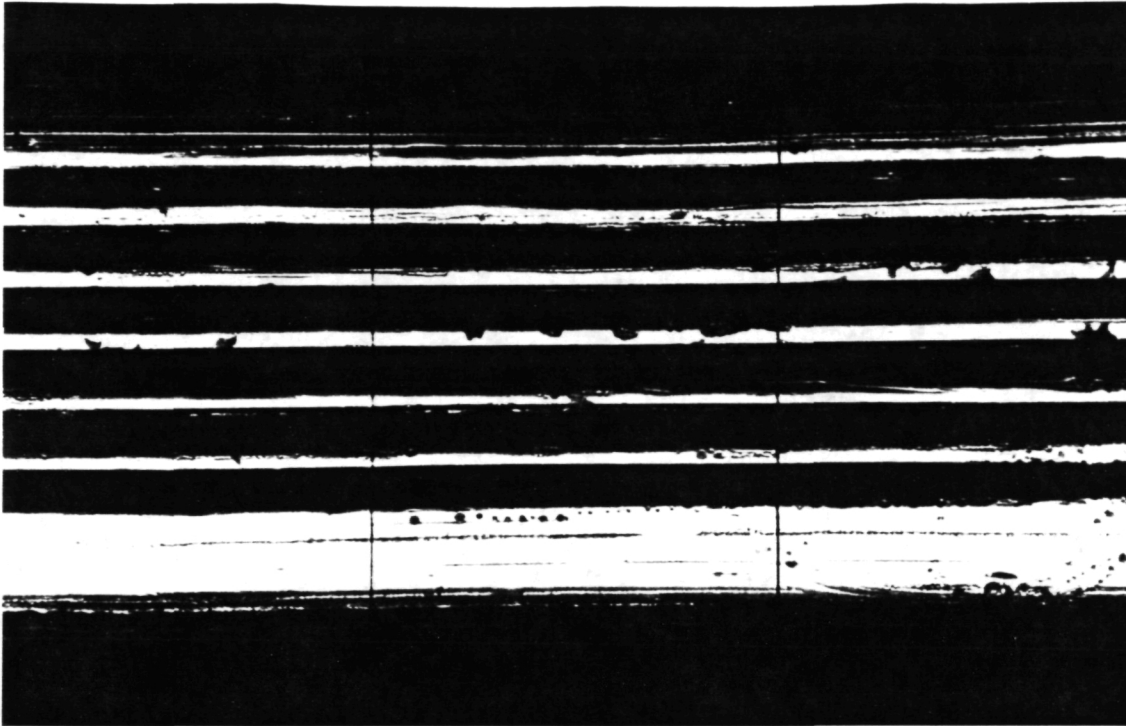


Figure 20. REGENERATOR SECTION, PLATE SPACING

Diaphragms inherently perform the function of sealing the positive displacement working volumes of the expander and compressor. This flexural sealing technique avoids rubbing wear and obsoletes high precision, close running "clearance seals", simplifying construction and promoting long life.

III.3.2 4 K Cooler Layout

The 4 K cooler layout includes two diaphragm expanders and two diaphragm compressors interconnected by a recuperative regenerator described in Section III.2. The expander heads share a common 4 K "deck", to which a load heater is attached for purposes of measuring cooling capacity. The compressor heads similarly share a common 20 K "deck" which is thermally connected to a 20 K heat sink provided by helium boil off and described in Section III.3.3. The compressor head "deck" and the expander head "deck" are suspended beneath a 300 K motor deck by thin walled tubes which provide alignment while limiting structural heat leak. The compressor and expander diaphragm hubs are each connected to mating diaphragm hubs on the motor deck by similar thin walled tubes which serve to transmit diaphragm stroking forces from motors located on the motor deck. This overall arrangement is illustrated in Figure 21.

Rotary motors drive the reciprocating compressor and expander assemblies through conventional kinematic crank shaft and connecting rod mechanisms, with the four motor deck diaphragms serving as cross heads. The cryocooler cold end assembly is found (photo inverted) in Figure 22, and the kinematic drive system components are displayed in Figure 23.

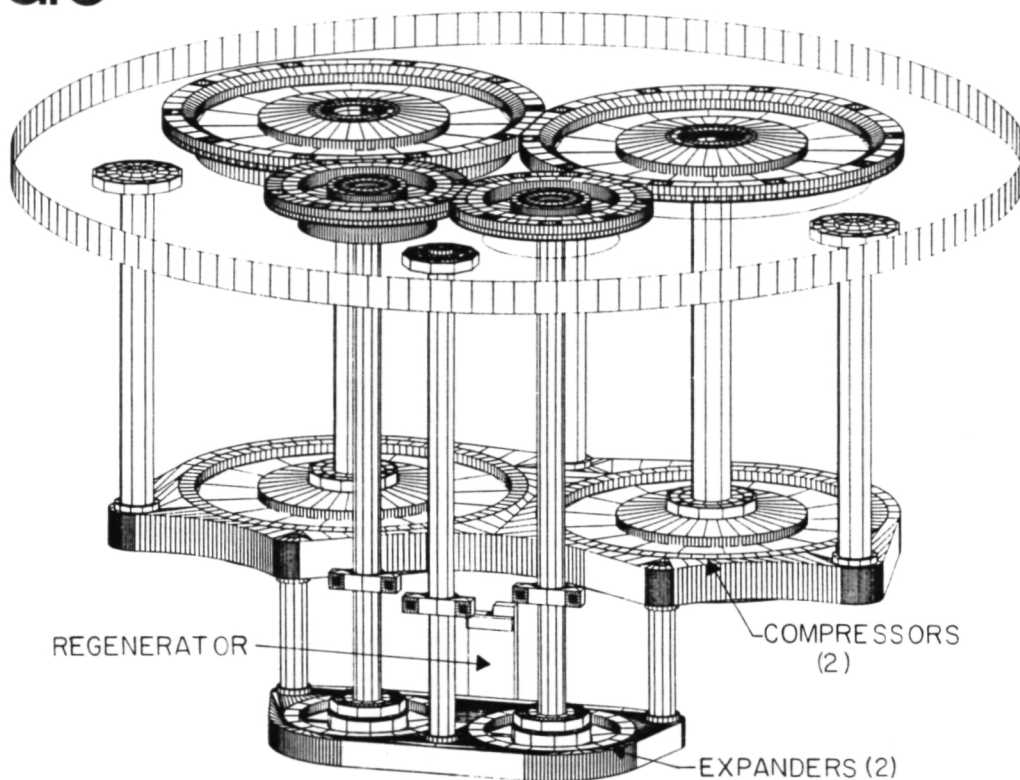


Figure 21. 200 mW 4-20 K COOLER ASSEMBLY

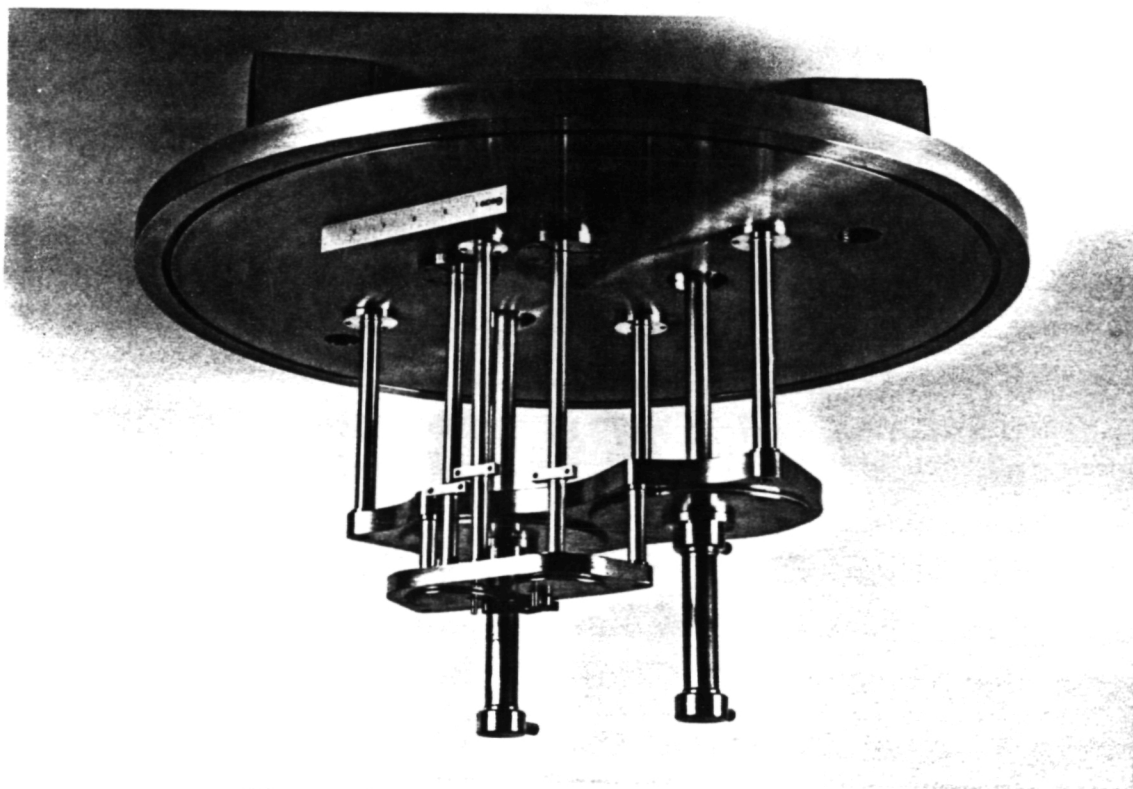


Figure 22. CRYOCOOLER COLD END ASSEMBLY

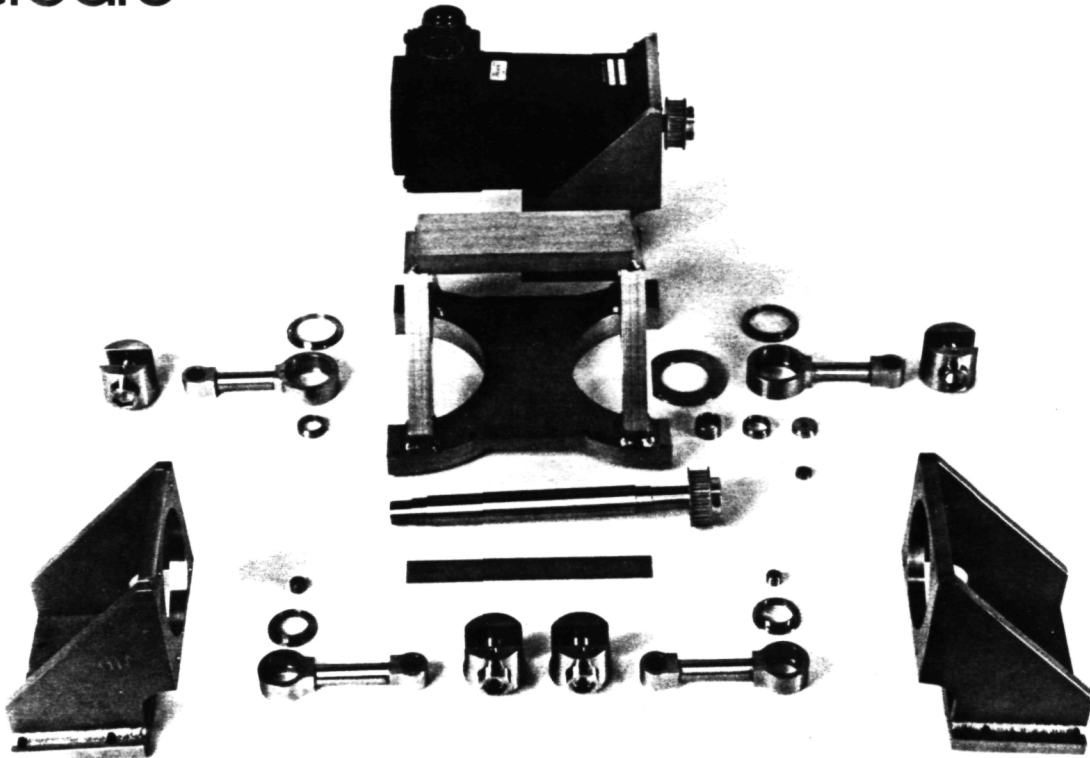


Figure 23. CRYOCOOLER DRIVE SYSTEM COMPONENTS

The cooler is fabricated primarily of Type 304 stainless steel, chosen for its cryogenic properties and relatively low thermal conductivity. Copper inserts are used where high conductivity is desired, such as for distributing heat from the load heaters to the expander deck heads and from the compressor deck heads to the heat sink heat exchangers. The working gas boundary is hermetically sealed by furnace brazed joints.

Construction of the 4 K cooler was completed up to the point of machining head contours to match the deflected and pressure loaded diaphragm contours before funding limitations forced suspension of fabrication activities. Components were subsequently placed in controlled storage to enable completion of fabrication when new funding is secured.

A complete drawing package defining the 4 K cooler and its individual components is provided as part of the final report deliverable.

III.3.3 Facility Integration

The 4 K cryocooler was designed to integrate into a vertical bell jar vacuum system for testing. The 300 K motor deck sits atop a 24 inch diameter spool piece, which in turn rests upon the baseplate of the vacuum system. The spool contains numerous passthrough nozzles for instrumentation leads and plumbing connections to provide 20 K heat sinking. The zone defined by the motor deck, the spool, and the facility baseplate operates at vacuum levels between 10^{-6} and 10^{-5} torr, maintained by a diffusion pump backed by a rough pump. The

zone above the motor deck is covered by a 24 inch diameter bell jar, and this region is evacuated by the rough pump. This scheme precludes the potential for volatile contaminants from the motor and drive system bearing lubricants from fouling the cold surfaces of the cryocooler. At the same time, maintenance of nominal vacuum above and below the 24 inch diameter motor deck prevents structural deflections of the deck which would interfere with proper maintenance of diaphragm to head clearance. Spring loaded check valves installed in the motor deck and facing in both directions prevent any possibility of distorting the deck through improper sequencing of vacuum system controls during pump down. The 4 K cooler assembly is illustrated in Figure 24.

Experimental heat sinking from the 4 K cooler is provided by expendable helium cryogen. Copper heat exchangers of brazed construction are attached to the compressor head deck. Cold helium gas is supplied to the heat exchangers at the opposite end and flows upward through the exchangers toward discharge nozzles located near the deck end. Waste heat from the deck meanwhile conducts downward through the copper structure of the exchanger counter to the helium flow direction. In this fashion, the exchanger provides "counterflow" performance between a conductive heat stream and a convective coolant stream, utilizing the sensible cooling capacity of the helium gas between 4 K and 20 K while maintaining the compressor deck at 20 K. Helium flow control provides course temperature control of the compressor deck, and trim heaters on the deck provide fine control under experimental conditions. Heat exchanger construction is illustrated in Figure 25.

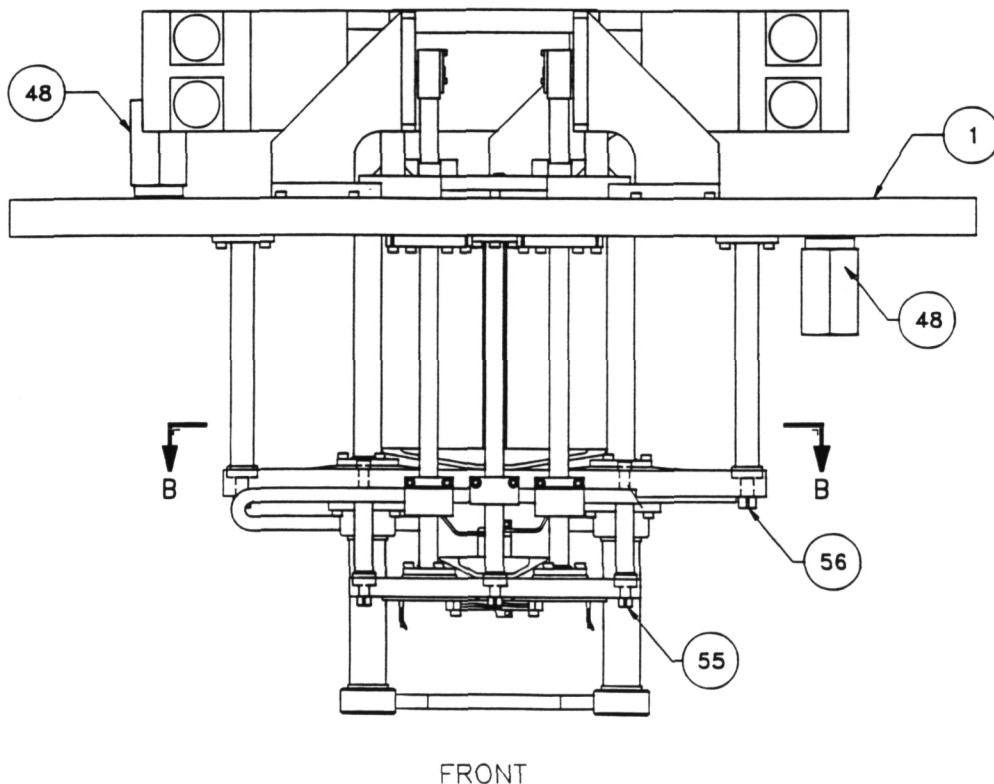


Figure 24. 4 K COOLER ASSEMBLY

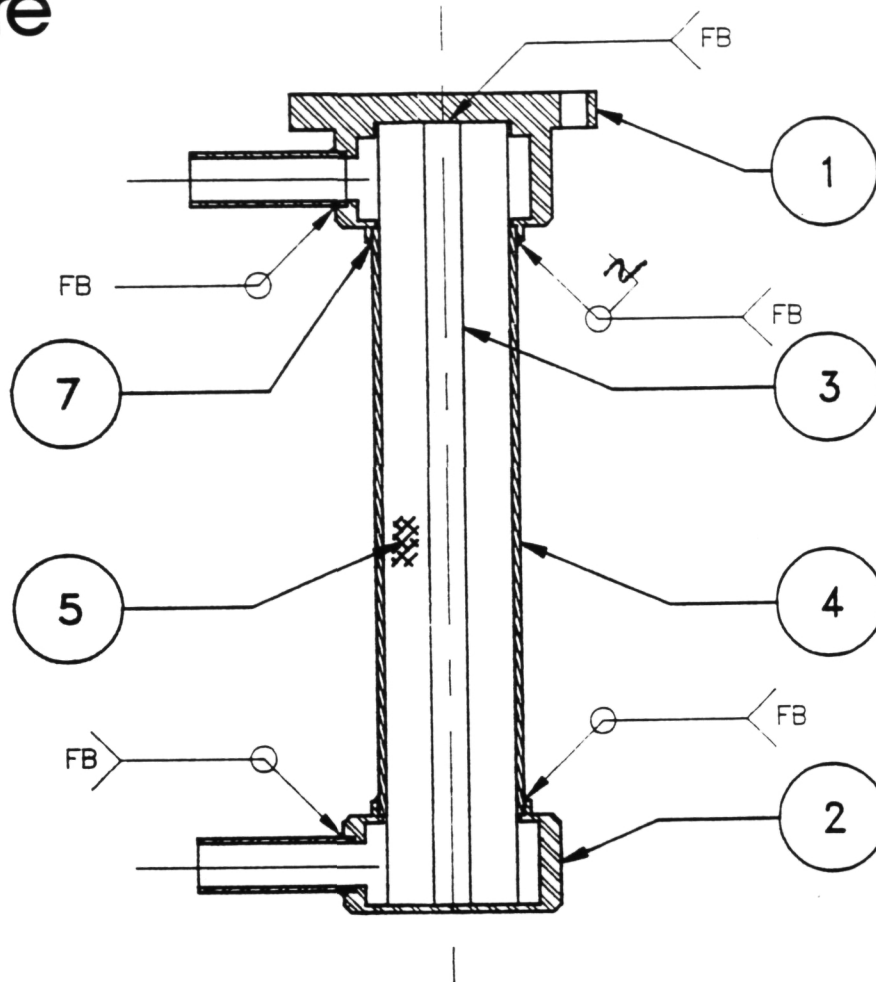


Figure 25. HEAT SINK HEAT EXCHANGER

III.3.4 Drive Motors and Controls

The drive motors are required to provide sinusoidal motion of the compressor diaphragm and "skewed sinusoid" motion wave forms for the expanders. This requirement is satisfied through the use of rotary DC drive motors and programmable closed loop feedback controllers, working through a kinematic linkage. One constant speed motor drives the two compressors at 180° phase angle through a common crankshaft, and two additional motors, one for each expander, provide the required programmable waveforms. Two separate expander motors are required because the expander waveforms are not symmetric, i.e., expander motion approximates a rounded sawtooth cycle. This cycle is simply achieved by programming the expander motor drives to vary rotational speed sinusoidally with the rotational position over each cycle of rotation. Amplitude of the programmed speed variation determines the degree of asymmetry in rising and falling slopes of the sawtooth position waveform. The kinematic drive consists of a simple eccentric rotary crankthrow driving the big end of a connecting rod through a roller bearing. The connecting rod wrist pin attaches to a long thin walled expander drive tube, the top end of which is supported by a locating diaphragm.

The drive motors are Pacific Scientific R-32 brushless DC motors with matched servo amps utilizing resolver feedback. These motors operate in a rough vacuum above the motor deck, while the cryocooler operates in a separately sealed, high vacuum region below the motor deck, to minimize the potential for structural deformation of the motor deck. Separate vacuum systems are employed for the two zones to eliminate any contamination potential from motor insulation or bearing lubricants.

A Creonics "MC" multiaxis position controller controls the Pacific Scientific servoamps to provide the required non-sinusoidal stroking of the two expanders and the sinusoidal stroking of the two compressors. This combination of components provides the required precision, speed, and ease of programming flexibility.

The process of specifying the motors included an assessment of their capability to follow a conservatively severe rounded sawtooth expander volume waveform, computed with our performance design code FORTIT to overshoot the requirement for regenerator balancing. We compared the rated torque of candidate motors to the computed polar moment of the motor rotor and the maximum drive torque requirement of the apparatus at the peak angular acceleration point in this conservative cycle to ensure an adequate operating margin.

Minor modifications were made to the motor shafts by machining to allow interface with the kinematic drive system components. Proper fit up of all mechanical components was verified before placement into controlled storage.

III.3.5 Diaphragm Fabrication

Significant development effort went into fabrication of suitably flat and stress-free stainless steel diaphragms for the 4 K cooler. The smaller expander diaphragms were ultimately successfully fabricated by furnace brazing of 10 mil thick stainless sheet stock between two symmetric stainless machined pieces, each comprising a rim half, a hub half and a structural bridge between the two which is machined away after brazing. The machined parts are stress relieved and lapped after machining to ensure retention of flatness during the subsequent braze cycle. Assembly flatness of better than two mils from the free state was achieved. This approach, which proved successful for the expander diaphragms, did not scale up successfully to the larger size of the compressor diaphragms. The machined components "oil canned" during stress relieving, and repetitions of the machining and stress relieving cycle failed to produce parts adequately flat to enable lapping. We were unable to successfully resolve difficulties in fabricating suitably flat compressor diaphragms within the funding available to this program. We have since, on a companion project, successfully demonstrated fabrication of next generation diaphragms by super plastic forming of 10 mil titanium sheet stock into a semitoroidal shape. This technique requires an investment in die tooling but reliably produces precision diaphragms with superior stress/strain characteristics and very low labor requirements. This advanced diaphragm design is a preferred substitute for the troublesome flat diaphragms unsuccessfully attempted under this program when additional funding is secured to resume and complete cryocooler fabrication and testing. The semitoroidal diaphragm is illustrated in Figure 26.

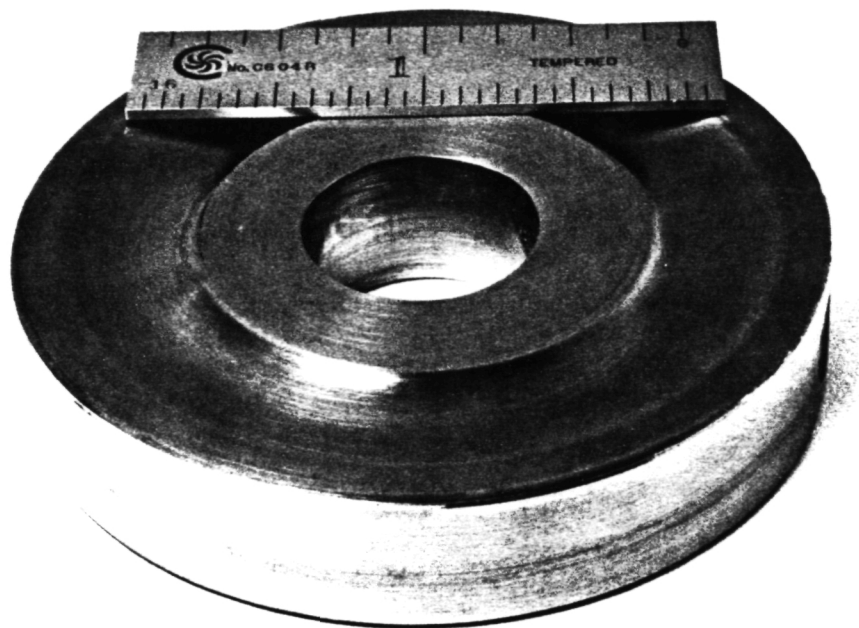


Figure 26. SEMI-TOROIDAL DIAPHRAGM ASSEMBLY

IV. CONCLUSIONS AND RECOMMENDATIONS

We conclude from Phase I experimental results and Phase II analytical results that practical and efficient refrigeration between 4 K and approximately 20 K is feasible through the use of a low pressure recuperative dual Stirling cycle approach. We further conclude that the low pressures necessary to successfully implement this approach require cascading the 4 to 20 K cooler beneath a more conventional 20-300 K high pressure refrigeration cycle to minimize system mass and volume.

We conclude, and have demonstrated, that it is feasible to fabricate miniature parallel plate heat exchangers suited to the requirements for 4 K cooling with ± 1 micron plate spacing precision and hermetic leak tight integrity.

We conclude that the performance and efficiency of Stirling cycle cryogenic refrigerators may be enhanced and extended by optimizing compressor and expander volume cycles on thermodynamic rather than kinematic criteria. Higher performance is possible as we move beyond constant speed crankshaft mechanisms and into programmable linear drivers.

Finally, we conclude that 4 K cryocoolers of very long operating life at high reliability and efficiency are feasible by means of diaphragm design concepts. These results are particularly pertinent to the cooling of sensitive scientific instruments aboard orbiting satellites and installed in remote unattended locations.

We strongly recommend that the Phase II hardware be completed and tested, based upon the successes and continued promise of the Phase II work. Carrying out this recommendation will require additional funding support and availability of the Phase II cryocooler hardware. Creare is actively pursuing additional funding to complete this effort and will continue to do so. We strongly recommend and request that NASA/JPL facilitate successful completion of this program's objectives by authorizing retention of the Phase II hardware at Creare for a reasonable time period. We recommend that a 2-year extension on hardware delivery requirements be granted to enable finding of additional funding and successful completion of the cryocooler fabrication and testing activities.

V. ACKNOWLEDGEMENTS

We gratefully acknowledge the support of NASA/JPL and the SBIR program in funding this research.

BIBLIOGRAPHY

1. Daney, D.E., *Refrigeration for Cryogenic Sensors and Electronic Systems*, NBS SP 607, (1981) 48.
2. Roubeau, Pierre M., *Refrigeration for Cryogenic Sensors and Electronic Systems*, NBS SP 607, (1981) 3.
3. Fleming, R.B.; Advances in Cryogenic Engineering; V.12, K.D. Timmerhaus, ed., New York, NY: Academic Press, 1967, Paper E-4, pp. 352-362.



Report Documentation Page

1. Report No. #8	2. Government Accession No.	3. Recipient's Catalog No.	
4. Title and Subtitle 4 K STIRLING CRYOCOOLER DEMONSTRATION FINAL REPORT		5. Report Date 9/22/92	
		6. Performing Organization Code	
7. Author(s) W. DODD STACY		8. Performing Organization Report No. TM-1563	
		10. Work Unit No.	
9. Performing Organization Name and Address CREARE INC. P. O. BOX 71 HANOVER, NH 03755		11. Contract or Grant No. NAS7-1111	
		13. Type of Report and Period Covered Final Report 1 May 1990 - 30 Apr 1992	
12. Sponsoring Agency Name and Address NASA-JET PROPULSION LABORATORY 4800 OAK GROVE DRIVE PASADENA, CA 91109		14. Sponsoring Agency Code	
15. Supplementary Notes			
16. Abstract This report briefly summarizes the results and conclusions from an SBIR program intended to demonstrate an innovative Stirling cycle cryocooler concept for efficiently lifting heat from 4 K. Refrigeration at 4 K, a temperature useful for superconductors and sensitive instruments, is beyond the reach of conventional regenerative thermodynamic cycles due to the rapid loss of regenerator matrix heat capacity at temperatures below about 20 K. To overcome this fundamental limit, the cryocooler developed under this program integrated three unique features: recuperative regeneration between the displacement gas flow streams of two independent Stirling cycles operating at 180° phase angle, tailored distortion of the two expander volume waveforms from sinusoidal to perfectly match the instantaneous regenerator heat flux from the two cycles and thereby unload the regenerator, and metal diaphragm working volumes to promote near isothermal expansion and compression processes. Use of diaphragms also provides unlimited operating life potential and eliminates bearings and high precision running seals. A Phase I proof-of-principle experiment demonstrated that counterflow regenerator operation between 77 K and 4 K increases regenerator effectiveness continued on back			
17. Key Words (Suggested by Author(s)) 4 K Stirling cryocooler Counterflow regenerator		18. Distribution Statement	
19. Security Classif. (of this report) UNCLASSIFIED	20. Security Classif. (of this page) UNCLASSIFIED	21. No. of pages 36	22. Price

16. Abstract

by minimizing metal temperature transient cycling. In Phase II, a detailed design package for a breadboard cryocooler was completed. Fabrication techniques were successfully developed for manufacturing high precision miniature parallel plate recuperators, and samples were produced and inspected. Process development for fabricating suitably flat diaphragms proved more difficult and expensive than anticipated, and construction of the cryocooler was suspended at a completion level of approximately 75%. Subsequent development efforts on other projects have successfully overcome diaphragm fabrication difficulties, and alternate funding is currently being sought for completion and demonstration testing of the 4 K Stirling cryocooler.